FINAL REPORT

CONTRACT NONR - 258 (00)

DEVELOPMENT OF A LIQUID ANNULAR RING TYPE

of

AIR COMRESSOR

To

OFFICE OF NAVAL RESEARCH

NAVY DEPARTMENT

Washington 25, D. C.

From

DEPARTMENT OF MECHANICAL ENGINEERING

School of Engineering and Architecture

HOWARD UNIVERSITY

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Prepared by Darnley E. Howard

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SUMMARY

A liquid annular ring type of air compressor composed of only two nontouching revolving parts which, nevertheless, act on the positive displacement principle, is designed, constructed, and its performance under operating conditions of pressure and gross vacuum investigated.

Final tests made over a range of shaft speeds, discharge pressures and volumes from 2000 to 3400 r.p.m., 5 to 15 lbs. per square inch, and 70 to 108 cubic feet per minute indicated a maximum isothermal efficiency of 70 per cent.

Although the efficiency of this simple low cost, smooth-running compressor exceeds that of the stationary annular liquid type, and compares favorably with large sizes of the centrifugal mass flow type, inherent losses seem to preclude its attaining the efficiency of the large axial flow type of air compressor.

INTRODUCTION

With the advent of the current gas turbine the problem of efficient compression of air in medium quantities has been emphasized. Piston compressors, while efficient, have serious drawbacks such as (1) limited capacity, (2) precision fits, (3) slow speed, (4) inertia effects and the like. On the other hand, mass flow compressors, such as the centrifugal and axial flow types, must be of high capacity to have a comparable efficiency. There is, then, a particular need for a non-reciprocating medium capacity compressor of high efficiency. Such a compressor is the liquid annular ring type which utilizes the centrifugal force of water to accomplish a higher compression in a given diameter than is possible with the centrifugal and axial flow types. It is to be further noted that when air under pressure is forced into the discharge port, this type compressor becomes a smooth running engine. Therefore, it appeared possible that this compressor engine combination could be competitive with the axial flow type gas turbine, especially in small sizes. Depending on displacement rather than high velocity mass flow to accomplish its compression, the annular ring type could use high heat conductivity hydrogen as a working medium in spite of its lightness.

Because of the possibilities envisioned above, this project was undertaken at the School of Engineering and Architecture, Howard University, Washington, D. C., under the sponsorship and with the financial assistance of the Office of Naval Research.

Several patents on devices of this kind have been issued, one of the earliest being No. 1, 038, 769, granted to Josef Lehne in 1912. Differing from this type in the manner of utilizing the liquid annulus is the Nash Jennings pump, originally patented in 1911 (No. 988, 133) and now being manufactured by the Nash Engineering Company of South Norwalk, Connecticut. In the latter compressor, the sealing liquid describes an elliptical annulus in a stationary casing.

A thorough search of the available literature has revealed very little experimental data on the performance of the rotating casing type compressor.

In a previous report entitled "Development of a Liquid Annular Ring Type of Air Compressor," May 1952, by Howard and Davis, data are presented on detailed studies of blade shapes, water movement in and around the rotor, partial sleeve bearings and tests of prototype No. 1.

The study of blade shapes consisted of varying the edge of the blades from sharp to round, or bulbous, and varying the angle of entrance into the ring of water. The phenomena were observed by

stroboscopic light and power measurements made. It was concluded that a slightly rounded edge on a straight (radial) blade of 1/16 inch thickness was best.

In the study of the partial sleeve bearing, a bench bearing test device of novel design was constructed. Driven by a carefully calibrated d-c motor, radial loads were applied by means of a lever system. The bearing rotated around a stationary shaft in which tubular probes were located at pressure points. These tubes led to pressure gages. As the load was increased the pressure on the gages increased. When this relationship ceased, i.e., when oil pressure failed to rise with load, failure of the bearing was indicated and the experiment was concluded.

After many modifications of bearing materials, oils, entering oil pressure, radial loads and speeds, it was concluded that a partial bronze bearing of wedge shape utilizing No. 10 S.A.E. oil would give a friction coefficient of 0.019 at 2500 r.p.m. and would therefore be feasible as an outboard bearing for the rotor. The coefficient for a conventional bearing was twice as great (0.039).

Prototype No. 1

A prototype was constructed utilizing the blades and bearing mentioned above. An isothermal efficiency of 63 per cent was

obtained at 3100 r.p.m. while delivering 36.7 cubic feet per minute at a pressure of 6 lbs. gage.

However, the sleeve bearing proved disadvantageous because of rapid wear, the need for an oil pump, and undue loss. It was, therefore, decided to design and equip future models with ball bearings.

Prototype No. 2

The second report (submitted May 1953) on this investigation described an advanced prototype which incorporated the following features:

(1) The rotor was completely ball bearing supported to minimize friction. (Figure 3)

(2) The blades were allowed a greater initial or constant submersion to prevent "blow by" of the air.

(3) Nine compartments instead of six or eight were provided to minimize the possibility of pulsing.

(4) The lap of the discharge and intake ports was increased to allow for re-expansion of clearance air and to accomplish greater compression in pockets before discharge.

(5) The ratio of eccentricity to diameter was increased to accomplish a longer submergence of blades.

This prototype was designed, built and given an extensive series of tests at different discharge pressures, speeds and volume. The results of the performance are presented for a range of pressures, speeds and volume.

APPARATUS

Air Compressor

In this air compressor a rotating annular ring of liquid, acting as a sealing and compressing medium (See Figure 1), is carried in a casing mounted on anti-friction bearings and eccentrically placed relative to a rotor with nine peripheral chambers. (See Figure 3.) On driving the rotor, air is trapped in these chambers, compressed and discharged through suitably placed ports in the tubular support. Revolved by the friction of the moving liquid, the casing rotates at 88 per cent of the speed of the rotor.

INSTRUMENTATION

Nozzle and Nozzle Tank

It was anticipated that this air compressor would deliver about 100 cubic feet of air per minute. In order to measure this volume accurately, a nozzle and nozzle tank were constructed according to dimensions specified by the A. S. M. E. Code, and shown on page 336 of Compressed Air Data Book, 5th edition, and published by the Compressed Air Magazine. The nozzle and tank are shown at "A" in Figure 5. Nozzle pressure was measured by a manometer (shown at "B") divided into 0.2 divisions.

Using the adiabatic approximate formula No. 4 as given on page 366 in the Data Book, curves were plotted for various temperatures showing the flow in cubic feet for a given manometer (inches of water) reading. (See Figure 6.)

Discharge pressure was measured by means of a mercury manometer calibrated in pounds per square inch and divided into 0.1 divisions ("C" in Figure 5.); r.p.m. was measured by strobotac ("D" in Figure 5) at the same time that torque readings were taken. A transmission dynamometer described in the Appendix and shown in Figure 2 was used for power measurement.

TEST PROCEDURE

Prototype No. 2

From the results of the water movement test, reported in May of 1952, it appeared that one of the important losses in the operation of this type of pump was due to the variation of air pressure within the pocket which caused the water to move in and out. To verify this supposition, two casings were fabricated; one, 12 inches in diameter and another 8 inches, the normal design size. Since the larger drum offered a greater area for water movement, the achievement of a higher efficiency after testing would prove the validity of the supposition.

The difficulty experienced at first in dynamically balancing the casing was overcome by counterbalancing with strips of lead, and lining with a thin layer of wax machined in situ. In operation, because of depth of the liquid annulus, the revolving rotor was unable to bring the casing up to speed; so a motor drive was installed to revolve the casing at varying speeds independent of the rotor.

The results were quite poor, maximum efficiency being only 54 per cent after the casing motor input was subtracted. The conclusion drawn then was that if water movement loss had been minimized, losses from other sources must have increased.

The pump was then assembled with the normal 8-inch diameter casing and an extensive series of experiments were performed in which

both the speed and discharge pressure were varied, The results showed a maximum efficiency of 71 per cent at a speed of 3200 r.p.m. and discharge pressure of 10 lbs. per square inch; 66 to 68 per cent at 3000 r.p.m. and 9.6 lbs. per square inch; and 64 to 68 per cent at 2600 r.p.m. and 6.6 lbs. per square inch.

The variations in efficiency may be explained on the basis of operating procedure. It was, of course, desirable to have the discharge air free of fine particles of water and mist. On the other hand, efficiency increased as clearance was reduced. But reducing clearance meant bringing the inner surface of the water annulus very close to the ports with the result that the rush of air tore particles of water off the surface. When this was discovered, arrangement was made to feed a small continuous stream of water (See "E" in Figure 5) into the intake to make up for the effluence of spray and the higher efficiencies were then consistently maintained.

It had been considered that, since a given efficiency was obtained with water, a better efficiency might be obtained with a denser liquid. This was predicated on the assumption that losses due to air and water movement, turbulence, and friction are relatively constant and independent of the liquid used. If then a heavy liquid were employed, it would be subjected to a greater centrifugal force and thus produce a higher discharge pressure. Since higher pressure means higher output, a net

gain in efficiency should result.

Carbon perchlorene, a liquid (cleaning fluid) with specific gravity of approximately 1.5 and a viscosity less than water, was selected. It is relatively nontoxic, noninflammable, and easily obtainable. In the first trial, the perchlorene evaporated as rapidly as it was poured in. The vapor pressure of perchlorene being relatively high, rapid evaporation was attributed to the absence of perchlorene vapor in normal air. To remedy this situation and to create conditions for saturated air, the nozzle was arranged so that the discharge exhausted into a container from which a flexible metal hose led to the intake of the pump. Even this provision was unsuccessful due perhaps to leakage of fresh air through the convolutions of the hose.

Water was then introduced into the casing with the thought that it would stay on the inside of the annulus and protect the mass of perchlorene from contact with the air. But the two liquids, under the influence of the terrific turbulence, formed a frothy viscous emulsion which was so persistent that it took two days to separate. No conclusion was reached on the basis of the above test since the liquid level of the perchlorene was never maintained, although a hurried reading showed an efficiency of 63 per cent.

As a final experiment, it was decided to determine the efficacy of this pump for vacuum production. To this end a throttle valve was

placed on the pump inlet and a test performed. The vacuum thus achieved varied with the r.p.m., the maximum being at the highest speeds. At 3010 r.p.m., a vacuum of 13 inches of Hg was produced, the discharge was 71 cfm and efficiency 53 per cent.

EVALUATION

The feasibility of the rotating annular liquid ring air compressor may be partially summarized in the following observations:

(1) The investigation proved that this device may be easily constructed with a minimum of precise parts; that it is inherently balanced, rotates smoothly and delivers a relatively large quantity of cool oil-free air for its size.

(2) A minimum of fifteen points of loss were discovered, some of which are quite important. (See Appendix.)

(3) It appears that the most significant loss is that due to the water movement which accompanies the compression of air in each pocket. This seems to be an inherent loss.

(4) The maximum efficiency of 71 per cent obtained is to be compared with about 60 per cent for the elliptical liquid annulus type; the 70 to 75 per cent of the centrifugal; the 80 per cent of the axial flow; and 85 per cent of the piston type.

(5) An increase in efficiency with an increase in size cannot be expected since this series has progressed from a 1/2 hp model to a 6 hp model with relatively little increase. Also, an increase in size accentuates other problems such as acceleration and deceleration of air entering the pockets.

(6) For applications of medium volume (higher than the piston type but lower than the mass flow type compressor) and relatively low pressures (20 to 30 lbs. per square inch), it shows promise.

ACKNOWLEDGMENT

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Shoop, Charles F., and Tuve, George L., Mechanical Engineering Practice, 4th ed., McGraw-Hill Book Co., New York, 1949.

Vallance, Alex and Doughtie, Venton Levy, Design of Machine Members, 3rd ed., McGraw-Hill Book Co., New York, 1951.

Patents

1911	988,133	Turbo-displacement Engine	L. H. Nash
1912	1,038,769	Rotary Pump	J. H. Lehne
1912	1,045,732	Turbo-displacement Engine	L. H. Nash
1918	1,262,533	Rotary Compressor	G. C. McFarland
1918	1,281,792	Rotary Compressor and Exhauster	J. Johnston
1925	1,527,339	Compressor	E. Wilson
1933	1,919,252	Air Compressor	W. W. Paget
1935	2,006,366	Rotary Compressor	W. W. Paget

APPENDIX

In the course of the tests on Prototype No. 2, difficulty was experienced in accurately measuring the input to the pump. The losses in the motor were determined by the "constant loss" method which involved some assumptions. When to this is added sparking at the commutator, slipping of the belt and errors in the electrical instruments, the uncertainties became too high.

A more direct approach was then considered and a torque measuring instrument, inherently having a high degree of accuracy and employing a simple type of indicating system, was devised and fabricated. (See Figure 2.)

This transmission dynamometer, utilizing a helical spring in torsion as the measuring means, was placed directly on the end of the pump shaft and the deflection of the spring was measured by the movement of a sharp aluminum pointer over a paper protractor cemented to the pulley. The indicating system was "stopped" and the torque angle read by means of a stroboscopic light which was also used to determine the r.p.m.

The deflection of the spring for the maximum load was 60 degrees. Readings could readily be made to a half degree. The load being constant, there was little pointer quiver. Because of

vibration, friction of the ball bearings was practically eliminated. The instrument was, therefore, quite sensitive and readings were easily reproducible. Even after months of use, the pointer, as to be expected, still returned to zero at no load.

The dynamometer's sensitivity was additionally shown by its ability to measure the 1/4 hp of windage loss (6 divisions) even though designed for 6 hp at 3200 r.p.m.

The successful functioning of this device permitted the accurate measurement of power directly at the place where it was applied, independent of uncertainties in electrical measurement or belt slippage and dependent only on the accurate calibration of the spring.

It had been hoped that a dead weight calibration of the spring would have been adequate but probably due to an unbalance in the spring itself, the zero shifted as much as 10° at high speed. Hence, it became necessary to calibrate it dynamically by means of a 10 hp electric dynamometer. The calibration curve is shown in Figure 4. It is a band of curves(1500, 2000, 2500, 2800, 3000, 3200 r.p.m.) substantially straight, the spacing varying by r.p.m.

Each curve is made by plotting transmission dynamometer angle in degrees against electric dynamometer scale reading for a particular speed. In order to obtain the horse power, another family of curves was made (not shown) in which electric dynamometer reading was

plotted against hp for the given speeds.

It is interesting to note that a Polaroid (Land) camera has been used to photograph the phenomenon and the action was stopped completely. If there is any needle flutter, it will photograph as a band and the center of this band would then be the true needle position.

Sources of Power Loss

1. Hydraulic losses (from 55 to 65 per cent of total loss)
   a. Slipping of water relative to casing
   b. Reciprocation of blades
   c. Relative motion between tip of rotor blades and water
   d. Blade shape
   e. Flow of water in and out of chambers due to air pressure
   f. Movement of water due to centrifugal causes (lower angular velocity of casing relative to rotor)
2. Mechanical losses (14 to 20 per cent of total loss)
   a. Friction of casing bearings
   b. Friction of rotor bearings
   c. Friction of casing against atmospheric air
3. Air losses (20 to 30 per cent of total loss)
   a. Friction of air in and out of chambers
   b. Leakage of air

4. Miscellaneous

   a. Cutoff - discharge port

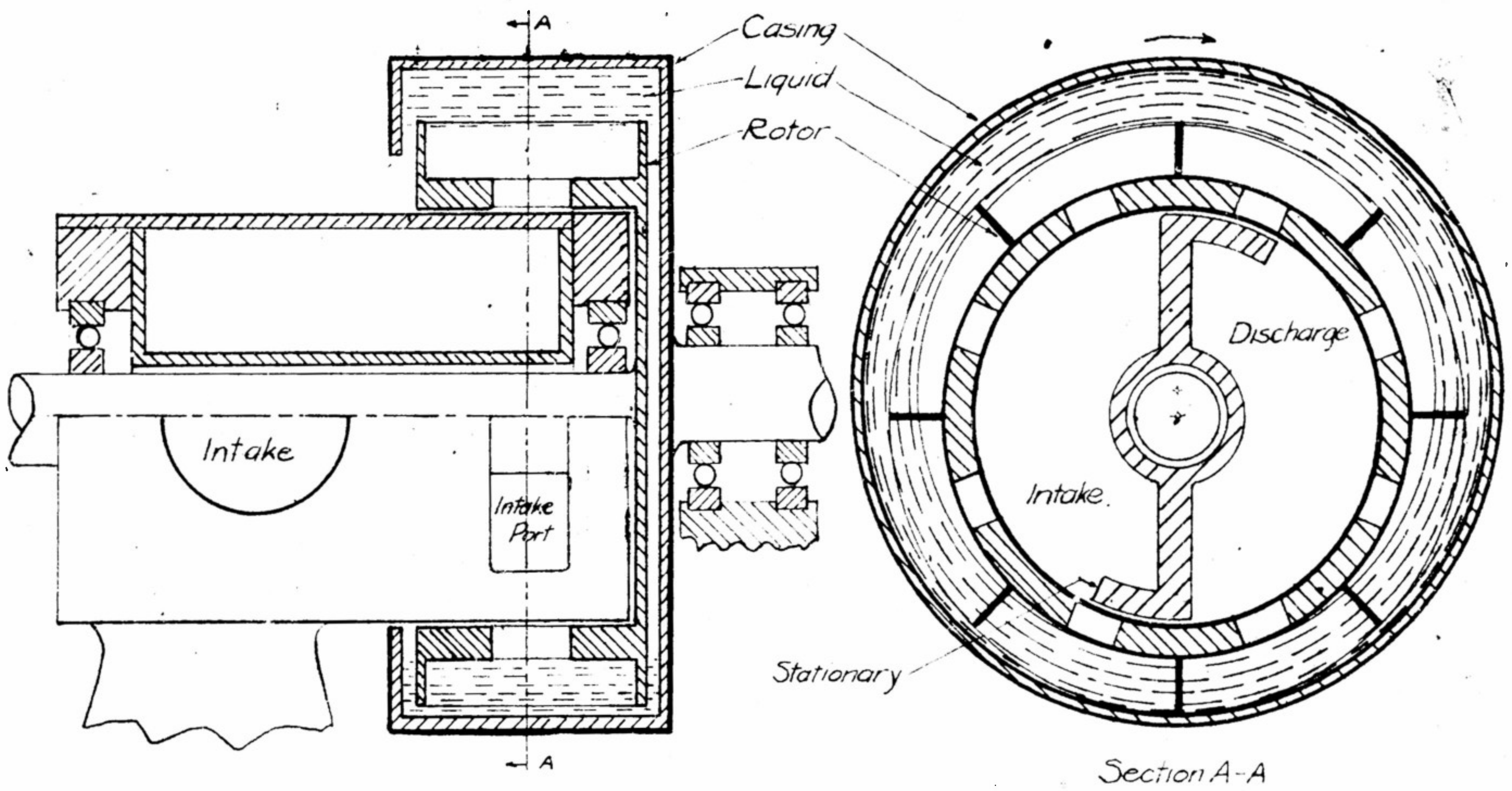
   b. Cutoff - inlet port

   c. Re-expansion of clearance air

TABLE 1. - PERFORMANCE DATA

Type of Test	Pump Speed (rpm)	Main disch. Pressure lbs/sq. in.	Nozzle Press. in H_2O	Torque in deg.	Suc. Press. In. of Hg.	Nozzle Temp.	Cubic Feet Free Air	Watts per cubic foot	Watts in discharge air	Watts Input	Eff.
Vacuum	3010	3.9	10.5	59	13	95	71	27	1930	3620	53
12-inch Casing	3100	5.7	a	a	a	a	68.5	16	1095	Net	54
Dense Liquid	2600	5.2	13.5	44	2.2	95	81	18	1460	2300	63
8-inch Casing With Water	3200	9.5	24.3	65	1.0	90	110	28	3080	4330	71.2

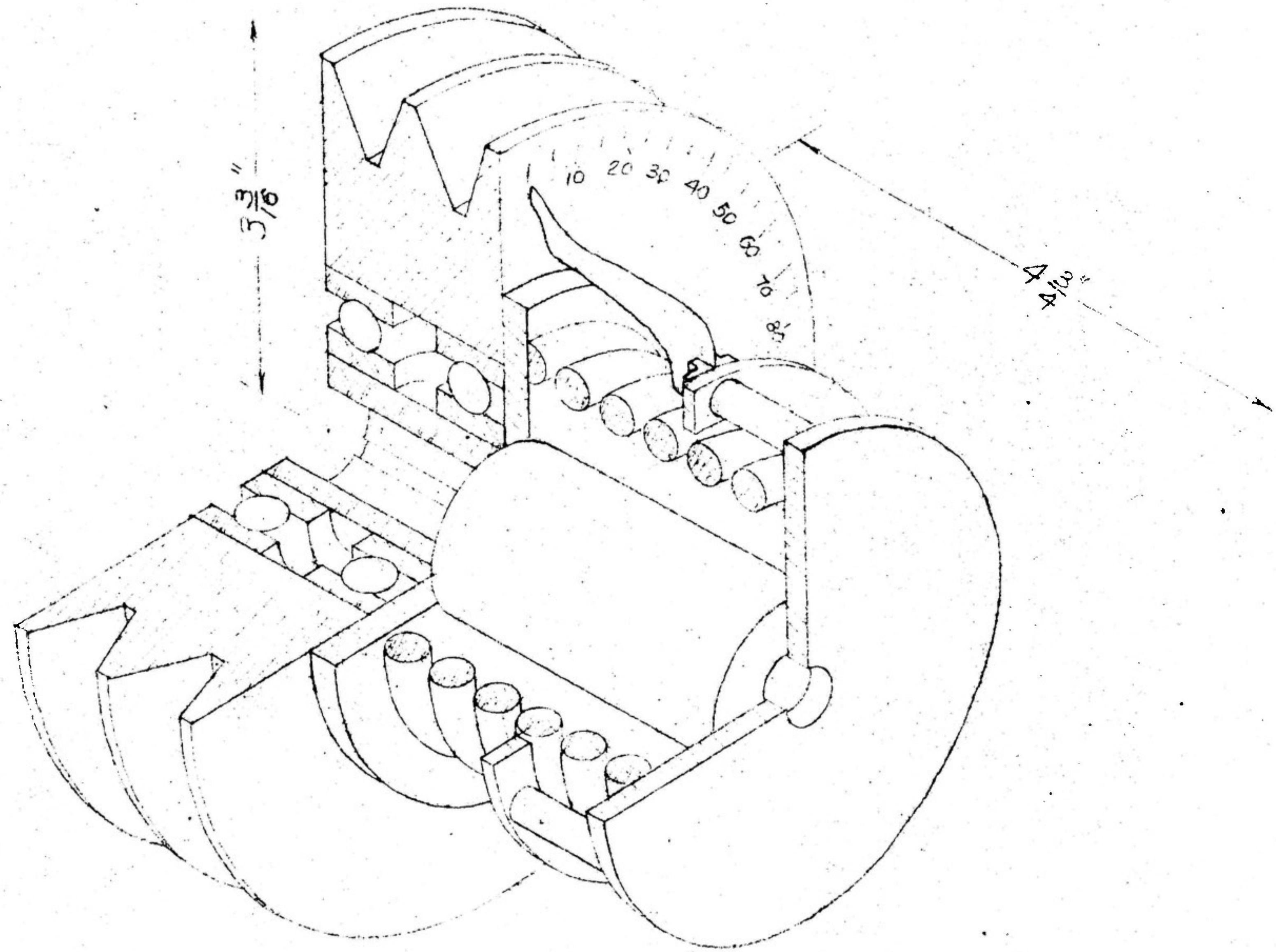
[a] Power input to rotor and casing was measured electrically and a 3/4-inch diameter nozzle used for air measurement.



ROTARY AIR COMPRESSOR

June 4. 1950 D.E. Howard

FIG. I



TRANSMISSION DYNAMOMETER

FIG. 2

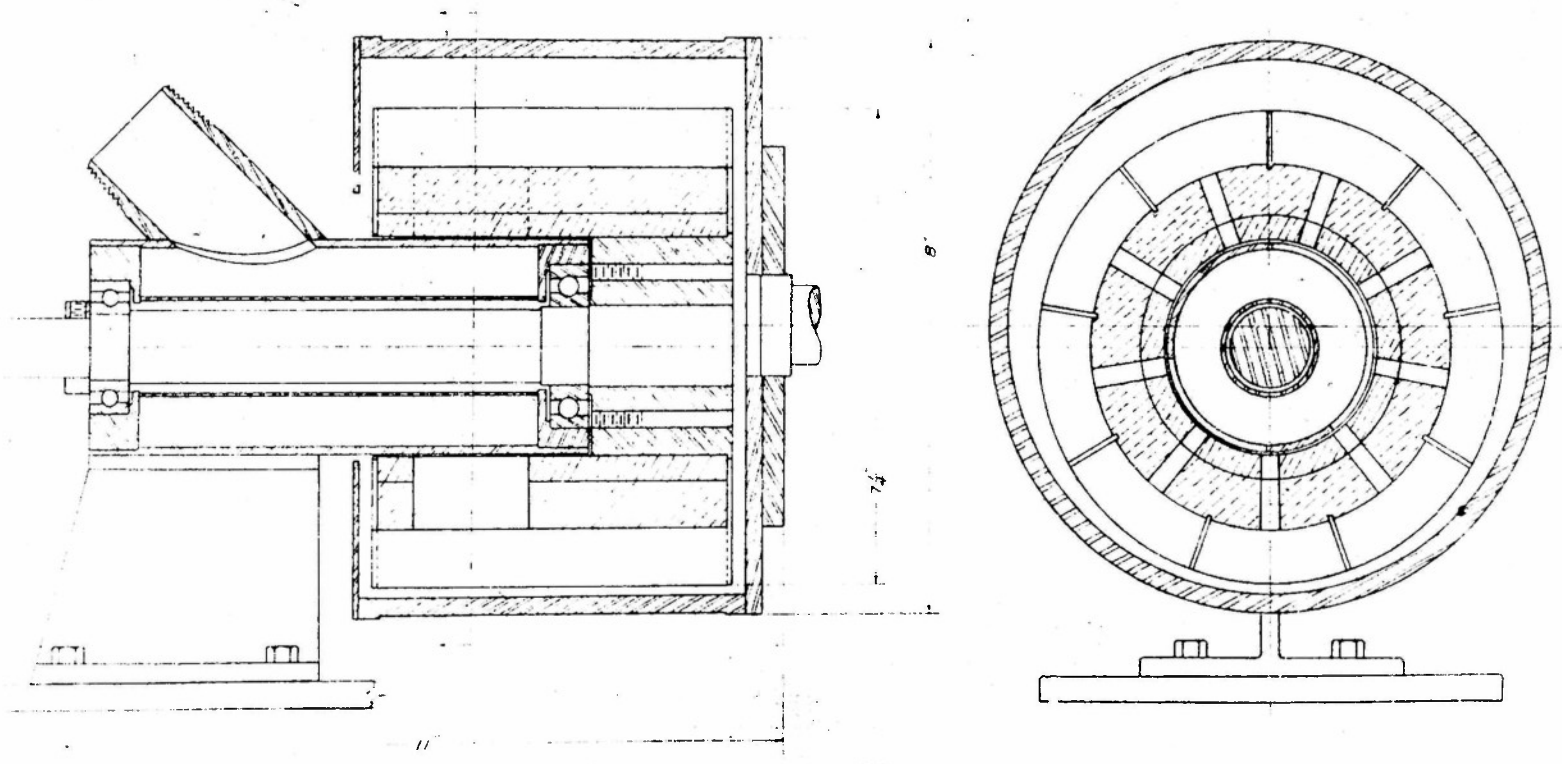


FIG 3

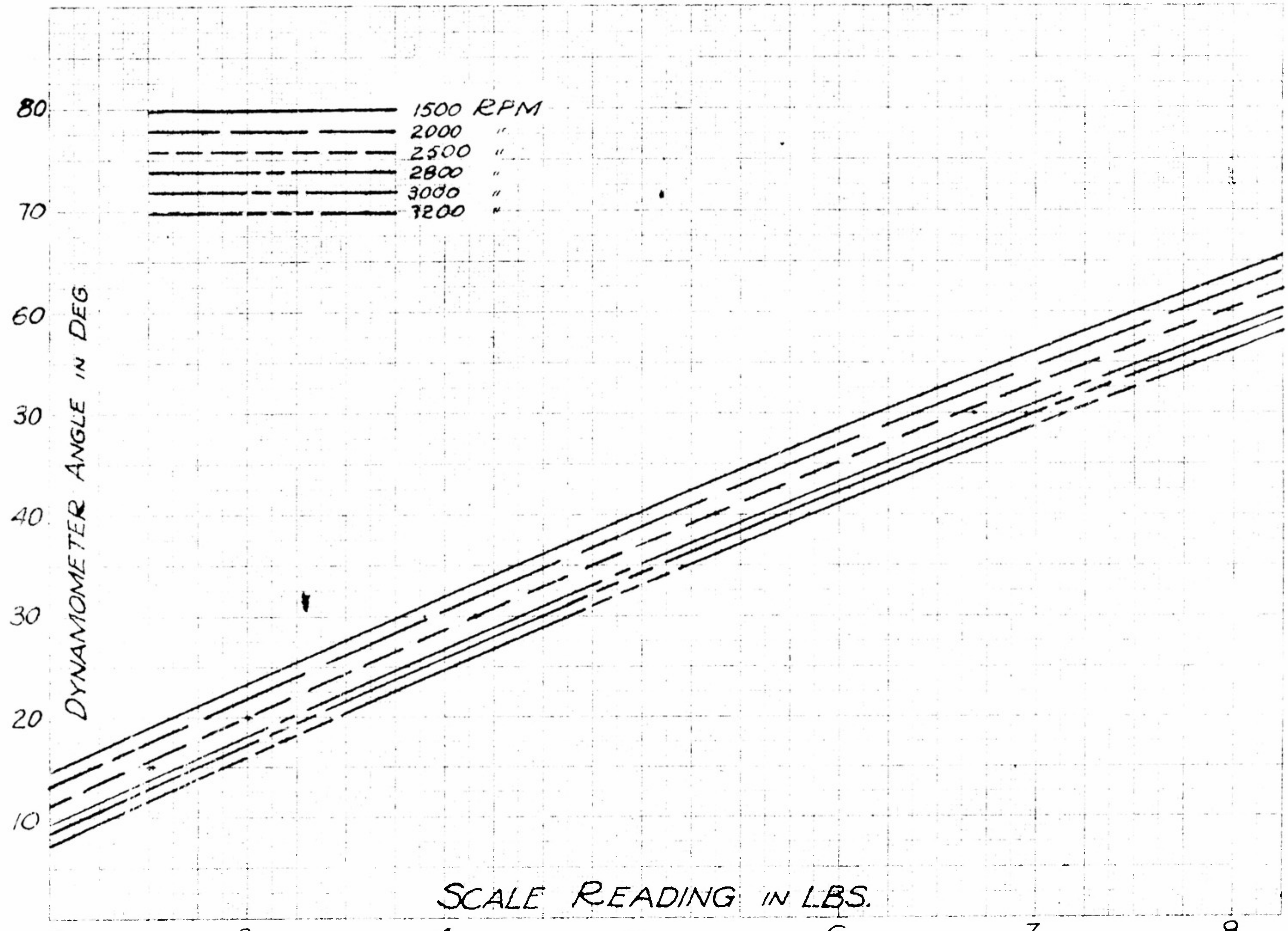
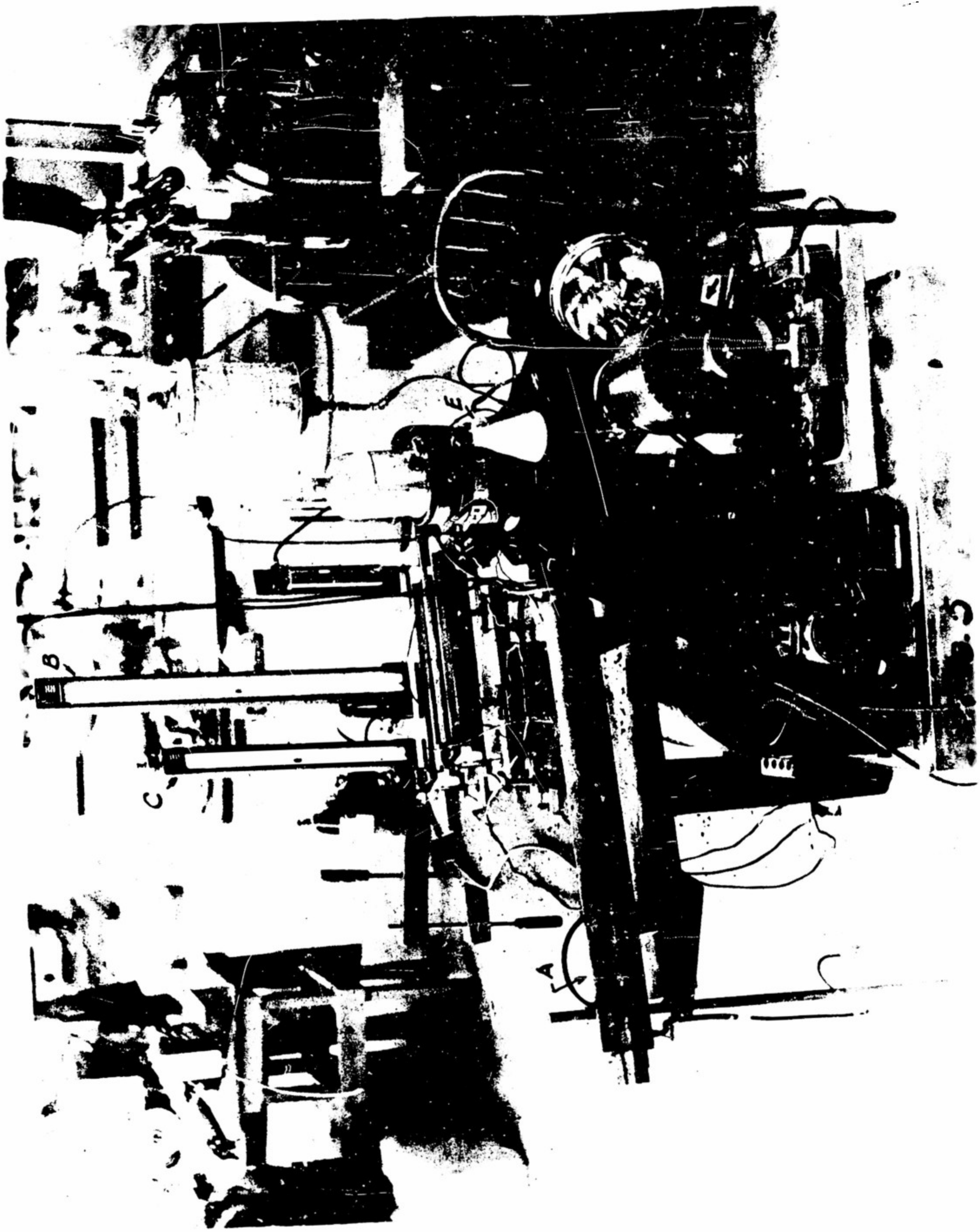
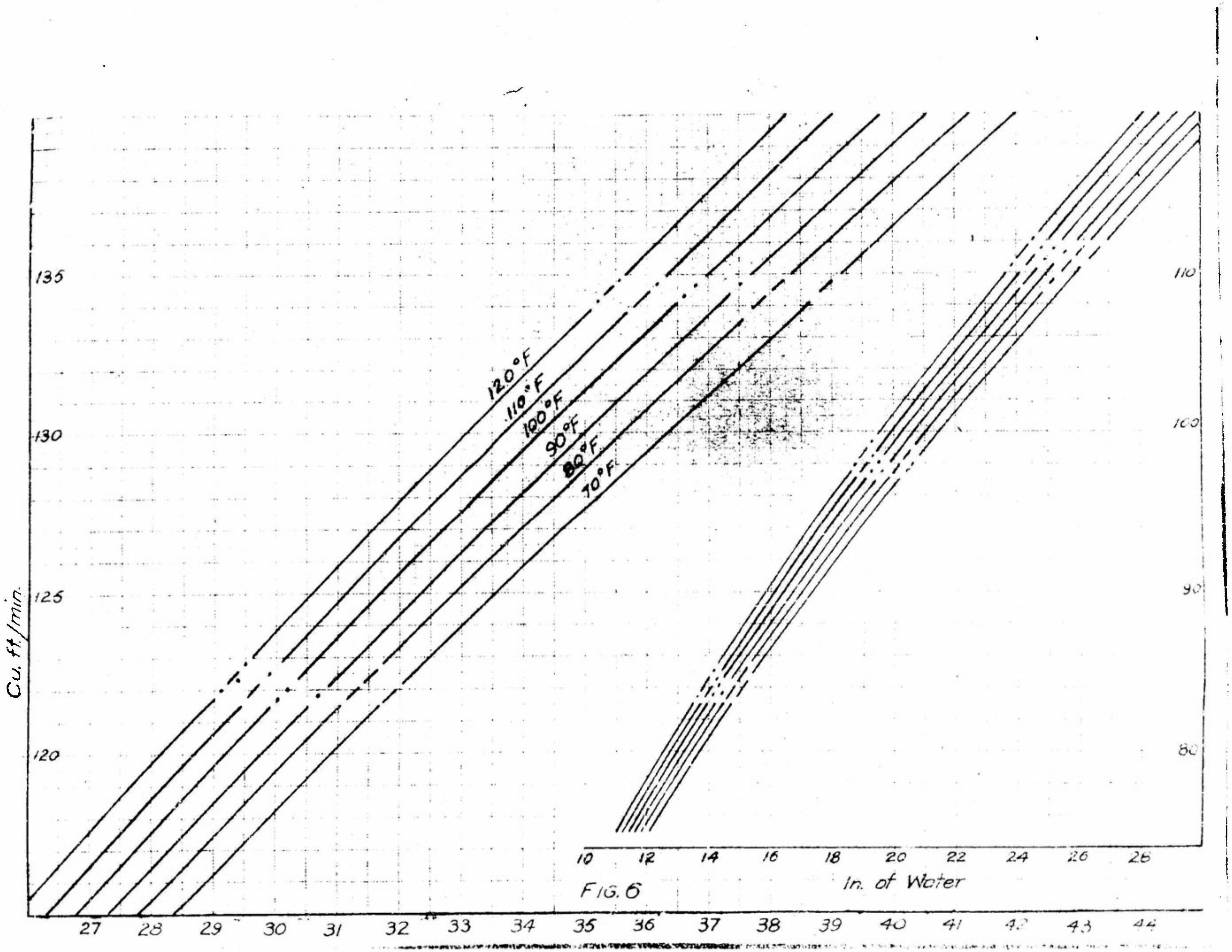

1500 RPM
2000 "
2500 "
2800 "
3000 "
3200 "
80
70
60
30
40
30
20
10
DYNAMOMETER ANGLE IN DEG
SCALE READING IN LBS.
2
3
4
6
7
8

FIG. 4



B
C
E
A



Cu. ft./min.
135
130
125
120
120°F
110°F
100°F
90°F
80°F
70°F
110
100
90
80
10 12 14 16 18 20 22 24 26 28
In. of Water
27 28 29 30 31 32 33 34 35 36 37 38 39 40 41 42 43 44

FIG. 6